TRANSLATION

INVESTIGATION OF GAS MOVEMENT AND HEAT TRANSFER IN ROTATING ROTORS

By

V. V. Mal'tsev

FOREIGN TECHNOLOGY DIVISION

AIR FORCE SYSTEMS COMMAND

WRIGHT-PATTERSON AIR FORCE BASE
OHIO

FTD-TT-63-366/1+2+4

UNEDITED ROUGH DRAFT TRANSLATION

INVESTIGATION OF GAS MOVEMENT AND HEAT TRANSFER IN ROTATING ROTORS

By: V. V. Mal'tsev

English Pages: 18

SOURCE: Russian Periodical, Vestnik Elektro promyshlennosti, 11, 1962, pp. 15-22

T-91
SOV/110-62-0-11-1-1

THIS TRANSLATION IS A RENDITION OF THE ORIGINAL FOREIGN TEXT WITHOUT ANY ANALYTICAL OR EDITORIAL COMMENT. STATEMENTS OR THEORIES ADVOCATED OR IMPLIED ARE THOSE OF THE SOURCE AND DO NOT NECESSARILY REFLECT THE POSITION OR OPINION OF THE FOREIGN TECHNOLOGY DIVISION.	PREPARED BY: TRANSLATION DIVISION FOREIGN TECHNOLOGY DIVISION WP-AFB, OHIO.

FTD-TT- 63-366/1+2+4 Date 11 June 1963
P

INVESTIGATION OF GAS MOVEMENT AND HEAT TRANSFER IN ROTATING ROTORS

V. V. Mal'tsev

(Candidate of Technical Sciences)

Investigations published recently [3] allowed us to explain the fundamental dependences and the significant influence of the rotational speed of the rotor of an electrical machine on the gas flow friction in channels.

In order to be able to determine the heat-transfer coefficients in axial and radial channels two experimental devices were especially constructed. The basic part of the device for investigating radial channels is a tube with an instrument head (Fig. 1) which is screwed into the cylinder on the shaft of the electric motor [3]. The air travels along the tube toward the axis of rotation or away from it.

Heat probe 6 (Fig. 1) is a small tube of fine glass cloth ribbon mounted in measuring cylinder 7 made of asbestos-cement material; to the inner surface of the small tube is attached a copper wire running zigzag coil to coil. The exposed part of the conductors is abraded and forms a continuous cylindrical copper surface divided by fine

insulation filaments. This surface is washed by a cooling flow.

The measuring cylinder was inserted and baked into an ebonite sleeve 2 which was then heat-insulated by asbestos linen with intermediate layers of aluminum foil 5.

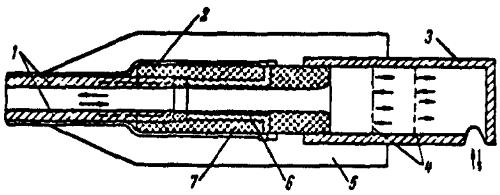

Fig. 1. Measuring head.
1) conductors; 2) ebonite sleeve; 3) cap; 4) screen; 5) heat insulation; 6) heat probe; 7) measuring cylinder.

The experiment was conducted in the following manner. The examined channel received the prescribed rotational speed and specific air flow. Then a certain constant current I was passed through the probe, heating its active surface. When a steady-state heat regime was attained we measured the rotational speed n, the velocity of the air in the channel v, the probe current and the voltage drop on the winding of the probe U; from the magnitude of the winding resistance of the probe R we determined the probe temperature and the dispersed power.

In order to determine leakage a special test was conducted during which the outlet from the tube was closed, the probe was heated, and we determined the dependence of the power required by the probe on the heating of its winding above the temperature of the ambient medium θ for various rotational speeds

$$P_1 = f(\theta).$$

When processing the test data on the power fed to the probe $P_p = UI = I^2R$ we calculated the power removed by the air $P_\alpha = P_p - P_1$ and then the heat-transfer coefficient:

$$\alpha = \frac{P_\alpha}{F\Delta t},$$

where F is the area of cooling of the probe, Δt is the temperature

drop between the wall and the air in the channel.

To determine the coefficient of heat transfer in the axial channels of a turbogenerator rotor and its dependence on the rotational speeds a device was constructed similar to one described earlier [3], the only difference being that the rotating cylinder on the shaft of the motor is more massive and the instrument head is placed axially in it.

Figure 2 shows a probe for determining the heat-transfer coefficient in the examined channel. The probe has an internal coil 1 onto which is wound a basic heating coil. Into the internal coil is placed insulation cylinder 2 with series-connected thermocouples for measuring the temperature drops between the inner and outer walls of the insulation cylinder.

Compensation coil 3 is used to prevent transfer of heat. By changing the current in the coil we can determine that state of the probe for which leakage of heat from the inner coil is equal to zero and all its power is directed only into the channels of the investigated section. The temperature of the channel wall is recorded by thermocouple 4 placed in it. The temperature of the cooling air is determined by thermocouple 5.

In order to determine the heat-transfer coefficient tests were made on a stationary cylinder with rotational speeds of 200, 400, and 600 rpm.

Figure 3 shows the results of investigating the coefficients of heat transfer in radial channels with various rotational speeds and movement of the air away from the axis of rotation; Fig. 4 shows the same for movement toward the axis of rotation.

As is evident from the derived dependences, during laminar flow with an increase in rotational speeds the heat-transfer coefficient decreases. To explain this phenomenon it is necessary to return to an examination of friction flow [3] obtained during an investigation of the movement of air in rotating radial channels (Fig. 7).

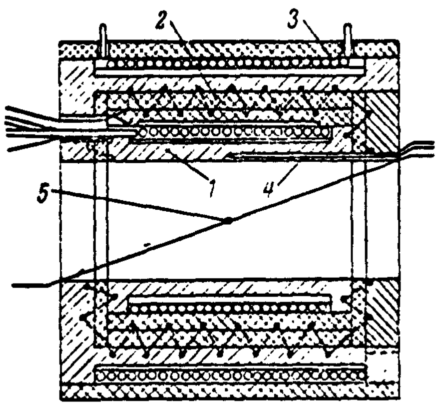

Fig. 2. Sensor for measuring the heat-transfer coefficient.

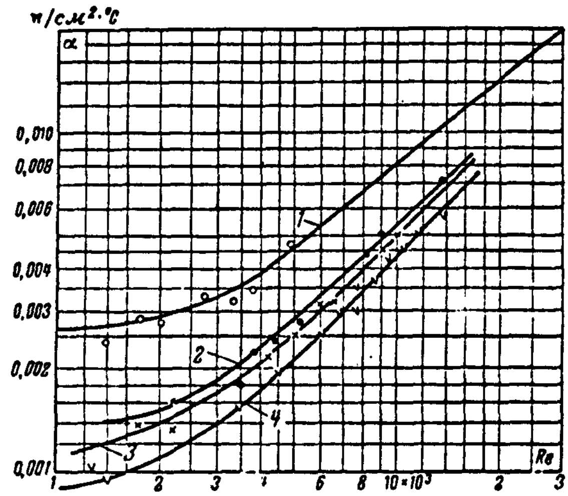

Fig. 3. Heat-transfer coefficients in radial channels of a rotor during centrifugal movement of air.

1), 2), 3), 4) when n is equal to 0, 500, 750, and 1000 rpm, respectively.

With an increase in rotational speed the coefficient of flow friction increases and the conversion to turbulent flow occurs at high Reynolds numbers; centrifugal forces damp the vorticity in the flow.

We will examine a particle of moving gas in a field of centrifugal forces; this particle, having received a disturbance from the wall or some other disturbed particle, should move in a radial direction toward the axis of rotation. Such a particle should change from a large radius of rotation to a small one, i.e., it should perform work, overcoming centrifugal forces which can be many times greater than

the forces disturbing the particle. Therefore, radial movement of a particle is rapidly stopped; pulsations in the direction opposite that of the centrifugal forces are damped under their action. The Coriolis forces exert a similar influence in a transverse direction. On account of this, the magnitude of critical Reynolds numbers becomes greater with an increase in the rotational speed.

The flow frictions of a laminar flow for the same Reynolds numbers increase with an increase of the rotational speed of the rotor [3], i.e., they become the same as they would be at lower Re in a fixed channel (with less disturbed flow). Along with this, the transfer of heat from the walls of the channel into the middle of the flow becomes less intensive with an increase in centrifugal forces.

Free convection also decreases on account of the difference of specific weights of hot and cold particles in the field of centrifugal forces on the walls of radially-oriented channels.

The smoothing out of vorticity in the field of centrifugal forces occurs both during movements toward the axis of rotation and without such movements. However, the magnitudes of the heat-transfer coefficients when moving away from the axis of rotation and toward it for the same Re do not coincide (Figs. 3,4), although their nature is identical. Let us note, however, that if the curves corresponding to transfer with movement away from the axis of rotation shift to the right then they will coincide. This is evidently due to the fact that with movement away from the axis of rotation the probe was located approximately 30 diameters from the inlet, while with movement toward the axis of rotation it was 3 diameters from the inlet.

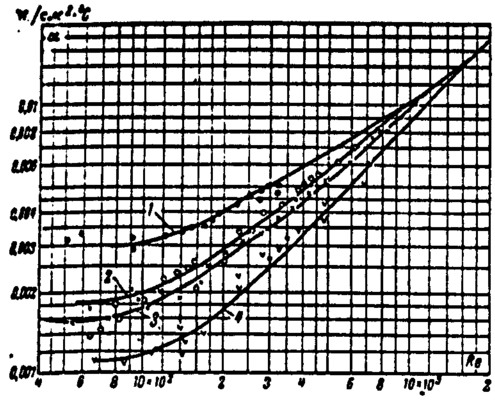

Fig. 4. Heat-transfer coefficients in radial channels with centripetal movement of the air. 1, 2, 3, 4) same as Fig. 3.

l/d	1	2	3	10	15	20	30	40	50
ε_l	1,9	1,7	1,44	1,28	1,18	1,13	1,05	1,02	1,0

It is known that the local coefficients of heat transfer α_{loc} decrease with distance from the channel inlet. If we take the ratio $\varepsilon_l = \frac{\alpha_{loc}}{\alpha}$ it will change with distance from the inlet as is shown in the table [1].

The results of the investigations of heat transfer in axial channels of a rotating rotor, given in Fig. 5, show that with an increase of the rotational speed during laminar flow the heat-transfer coefficients increase.

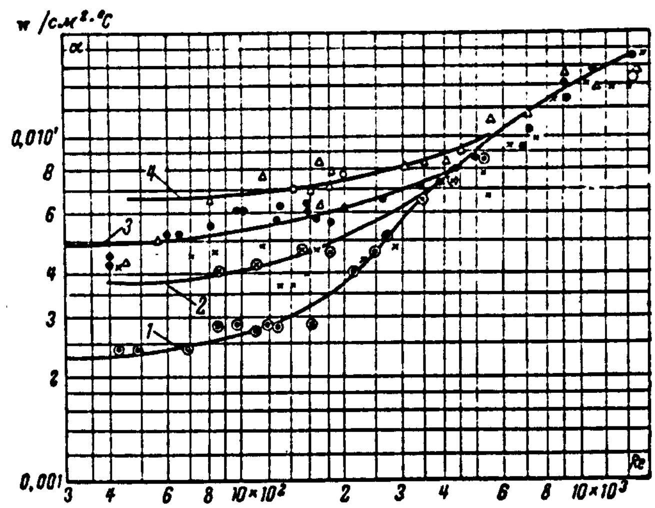

Fig. 5. Heat transfer coefficients in axial channels of a rotor. 1,2,3,4) when n = 0, 200, 400, and 600 rpm, respectively.

At first glance it seems paradoxical that in radial channels the heat-transfer coefficient decreases during rotation while in axial channels it increases, Upon closer examination of the vorticity in the axial channels this apparent contradiction can be explained.

The examined channel of circular cross section can be considered to consist of two halves: upper and lower. The centrifugal forces from the lower part are directed toward the upper; the boundary layer

at the walls of the channel are heated more than in the central part. In the upper half of the channel the colder layers of air, which are heavier, develop greater centrifugal forces than the light layers and, therefore, the cold air forces the heated air from the upper wall creating conditions of improved heat transfer. In the lower half opposite phenomena are observed. Such a distribution of the layers of air restrains vorticity, impairing heat transfer. Since the upper half of the channel is on the large radius with higher centrifugal forces, the effect of improving heat transfer in the channel prevails.

Besides this, the expelled heated layers move along the walls creating transverse circulation in the boundary layer which spreads to the lower half of the channel, improving heat-transfer conditions. Thus, the total effect during laminar flow in an axial channel is characterized by the increased heat-transfer coefficient with increased rotational speed.

In all cases of developing turbulent flow the heat-transfer coefficients for rotating channels correspond to fixed channels and can be determined by the usual formulas.

Heat transfer in the channel of a rotor is expressed by the following function:

$$P_к = F_к \alpha \Delta t, \qquad (1)$$

where F_k is the heat removal surface;

Δt is the average excess of temperature of the copper rotor over the temperature of the gas in the channel;

$$\alpha = \frac{\lambda_f}{d} \mathrm{Nu}, \qquad (2)$$

α is the heat-transfer coefficient;

λ_f is the heat conduction of the cooling gas;

d is the hydraulic diameter of the channel;

Nu is the Nusselt number.

For laminar movement in fixed channels

$$Nu = 0{,}17\,Re^{0.33}\,Pr_f^{0.43}\,Gr^{0.1}\left(\frac{Pr_f}{Pr_w}\right)^{0.25} \qquad (3)$$

where Pr_w and Pr_f are the Prandtl numbers in the boundary layer and middle of the flow respectively;

$Gr = \frac{gd^3}{v^2}\beta\Delta t'$ is the Grashof number;

v is the kinematic viscosity;

g is the acceleration of gravity;

β is the coefficient of cubical thermal expansion of gas;

$\Delta t'$ is the temperature excess of the wall over the temperature of the gas in the channel.

For turbulent movement

$$Nu = 0{,}021\,Re^{0.8}\,Pr_f^{0.43}\left(\frac{Pr_f}{Pr_w}\right)^{0.25} \qquad (4)$$

or for diatomic gases

$$Nu = 0{,}018 \cdot Re^{0.8}. \qquad (5)$$

As is shown by calculations and comparisons of the coefficients of heat transfer found experimentally we should introduce, a correction factor into Formula (3), which takes into account the dimensions of the channel, the rotational speed, and velocity of the gas in the channels:

$$\varepsilon_{pr} = \frac{1}{\sqrt{\frac{\omega d}{v_k} - 1{,}4 \cdot 10^{-4}\,Re}}, \qquad (6)$$

where ω is the angular rotational speed of the channel;

v_k is the relative gas velocity in the channel.

Then the heat-transfer coefficient for a rotating radial channel with laminar movement of a fluid (gas) is

$$\alpha_{pr} = \varepsilon_{pr}\alpha, \qquad (7)$$

where α is defined by Formula (2):

$$\alpha = 0{,}17\,\varepsilon_{pr}\,\frac{\lambda_f}{d}\,Re^{0.33}Gr^{0.1}Pr^{0.43}\left(\frac{Pr_f}{Pr_w}\right)^{0.25} \qquad (8)$$

The heat-transfer coefficient in rotating axial channels during laminar flow is

$$\alpha_{pa} = \varepsilon_{pa}\alpha, \tag{9}$$

where

$$\alpha_{pa} = \sqrt{\frac{\omega d}{v_{\kappa}}} - 1.4 \cdot 10^{-4} Re. \tag{10}$$

In order to refine and generalize Expressions (6) and (10) into a single calculation formula it is necessary to conduct physical investigations to determine the increase, the thermal boundary layer, and the peculiarities of vorticity.

The available data allow us to formulate four oonditions which determine the peculiarities of heat transfer in rotating channels:

1) During laminar flow in the channels of a rotating rotor the coefficients of heat transfer depend on the rotational speed.

2) The coefficients of heat transfer in rotating radial channels during laminar flow decrease when the rotational speed increases.

3) The coefficient of heat transfer in axial channels during laminar flow increases with increasing rotational speed.

4) With developed turbulent flow the coefficients of heat transfer in the rotating channels do not depend on the rotational speed.

Let us examine the movement of a gas flow in the air gap of an electrical machine when the rotor rotates with a periphal velocity u and the gas moves along the gap with velocity v_a.

Under the effect of the force of pressure in an axial direction

$$P_a = \Delta H F_3, \tag{11}$$

where F_z is the cross-sectional area of the air gap;

ΔH is the pressure drop between the inlet to the gap and the outlet from it.

The gas flow in the gap moves helically with an absolute velocity c which with respect to the surface of the stator will be expressed as $c_s = \sqrt{v_a^2 + v^2}$, and relative to the rotor $c_r = \sqrt{v_a^2 + w^2}$. (here v_a is the axial velocity component; v and w

are the tangential components relative to the stator and rotor, respectively).

The total force of the reaction of the surfaces of the rotor and stator P is proportional to c^2.

When calculating electrical machines we are usually interested not in this force but in two components: the tangential component P_τ on the rotor, since it determines the moment of resistance to rotation of the rotor and the associates energy loss to friction against the gas, and the axial component P_a which determines the pressure losses of the fan which ensures gas flow along the gap.

In the given case an axial force is examined which is equal to the sum of the axial components on the surface of the stator P_{as} and on the surface of the rotor P_{ar}:

$$P_a = P_{as} + P_{ar}; \quad P_{as} = P \frac{v_a}{c_s} = \tau s \; Fs \frac{v_a}{c_s}; \quad P_{ar} = P \frac{v_a}{c_r} = \tau_r F_r \frac{v_a}{c_r};$$

$$P_a = C_s \frac{\rho c_s^2}{2} F_s \frac{v_a}{c_r} + C_p \frac{\rho c_r^2}{2} F_r \frac{v_a}{c_r} = C_s F_s \frac{\rho c_s v_a}{2} + C_r F_r \frac{\rho c_r v_a}{2} \qquad (12)$$

where $\tau_s = C_s \frac{\rho c_s^2}{2}$ and $\tau_r = C_r \frac{\rho c_s^2}{2}$ are the tangent stresses on the surface of the stator and rotor, respectively.

Having substituted P_a from (11) into (12) and assuming that

$$F_3 = \pi (R_2^2 - R_1^2);$$
$$F_s = 2\pi R_2 l;$$
$$F_r = 2\pi R_1 l;$$
$$\delta = R_2 - R_1,$$

we obtain:

$$\Delta H = C_s \frac{l}{2\delta - \frac{\delta^2}{R_2}} \rho c v_a + C_r \frac{l}{2\delta + \frac{\delta^2}{R_1}} \rho c_r v_a. \qquad (13)$$

When $C_s = C_r = C$ and $\delta \ll R$

$$v=\frac{u}{1+\frac{R_2}{R_1}\sqrt{\frac{c_s}{c_r}}}\approx\frac{u}{2};$$

$$w=u-v=\frac{u}{2};$$

$$c_r=c_s.$$

having substituted in (13)

$$C=\frac{\Delta H}{\frac{l}{\delta}\rho c v_a}.$$

If we express C by means of the dynamic head of velocity $\sqrt{cv_a}$ and assume $\lambda = 2C$ then similarly with normal channels

$$\lambda=\frac{\Delta H}{\frac{l}{\delta}\frac{\rho c v_a}{2}} \qquad (14)$$

In order to experimentally determine the coefficient of resistance of an air gap a device was constructed whose basic part is a rotating rotor inserted into an interchangeable cylinder.

At one end the gap between the cylinder and the rotor communicates with the chamber; the other end of the gap communicates with the laboratory. Attached to the chamber is a fan which ensures the admission and feed of air into the tube in which a multiplier measures the air flow through the device. The rotational speed of the rotor is varied from 0 to 4000 rpm.

The velocity field between the rotor and cylinder in planes along the rotor is measured by a microtube and an electrothermoanemometer. The device includes four interchangeable cylinders having various diameters of internal bore.

In order to determine the effect of roughness of the stator and rotor on the coefficients of friction C_r and C_s the surfaces of the latter were covered with emery cloth of various grit, smooth surfaces (tracing paper), and also corrugations in the form of a wound spiral with different pitch and with different magnitudes of the projection.

Figure 6 shows the results of the experimental investigations processed using Formula (14).

As is evident from Fig. 6, if for the determining speed we assu $\sqrt{cv_a}$ and for the determining dimension $R_2 - R_1 = \delta$ (see line P anu the experimental points near it), the magnitude λ in the area of laminar flow coincides with Poiseuille's law. But the transition from laminar to turbulent flow occurs with Reynolds numbers which are lower than in fixed straight channels.

The conducted investigations and an analysis of the data [5,8] allow us to establish the following:

1) The values of the critical Reynold's numbers Re_{cr} depend on the rotational speed of the rotor, gas flows through the gap, geometri dimensions of the rotor and the gap, and the roughness of the surfaces of the rotor and stator, and is found within the limits

$$Re_{cr\,Q=0} < Re_{cr} < Re_{cr\,n=0}, \tag{15}$$

where $Re_{cr\ Q\ =\ 0}$ is the critical Reynold's number for a rotating rotor and in axial flow through the gap;

$Re_{cr\ n\ =\ 0}$ is the critical Reynold's number for a fixed rotor and axial flow through the gap.

2 The greater the ratio of axial velocity to peripheral velocity the larger the value of Re_{cr}.

3 With identically smooth surfaces of the rotor and stator and no axial flow, according to [5],

$$Re_{cr\,Q=0} = 41{,}3\sqrt{\frac{R}{\delta}}. \tag{16}$$

In straight smooth channels $Re_{cr\ n\ =\ 0} = 2200$.

Investigations, the results of which are given in [7-9], show that with axial flow the values of Re_{cr} do not exceed the limits set by Expression (15).

The majority of electrical machines with axial air flow through the gas have Re > 3000 i.e., the gas flow in their air gap is turbulent.

In the region of turbulent flow the coefficients of frictional resistance in an air gap with a rotating rotor correspond to the coefficients in an annular gap with a fixed rotor [see line B, Fig. 6 and its surrounding points as well as line 2 calculated by the Altschul formula $\lambda = 0.1\left(\varepsilon + \frac{100}{Re}\right)^{0.25}$.

If the surfaces of the stator and rotor have unequal roughnesses, the pressure drops are determined by Formula (13). The coefficients of resistance C_s and C_r are found from the expressions

$$C_s = 0{,}05\left(\varepsilon_s + \frac{100}{Re_s}\right)^{0.25}; \quad (17)$$

$$C_r = 0{,}05\left(\varepsilon_r + \frac{100}{Re_r}\right)^{0.25}, \quad (18)$$

where

$$\varepsilon_s = \frac{k_s}{\delta}; \quad \varepsilon_r = \frac{k_r}{\delta};$$

$$Re_s = \frac{\sqrt{c v_a \delta}}{\nu}; \quad Re_r = \frac{\sqrt{c_r v_a \delta}}{\nu}.$$

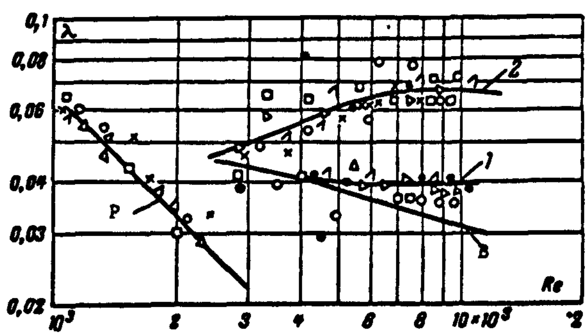

Fig. 6. Flow friction of an air gap with a rotating rotor.
1) rotor smooth, stator rough, ε = 0.08; P) from Poiseuille's law; B) from the Blasius law; ●) n = 0; ×) n = 500; Δ) n = = 1000; □) n = 2000; O) n = 3000, and Λ) n = 3500 rpm.

Figure 7 shows the results of the investigation of the flow friction of a gap with a corrugated rotor. The surface of the rotor has annular roughness with projections commensurate with the value of the gap, for which ε > 0.2 and ε < 0.2.

As is evident from Fig. 7, the calculated and experimental values coincide.

...the rotor is corrugated with an annular roughness commen-urate with the magnitude of the air gap $\varepsilon > 0.2$, the coefficient of resistance is analogous with labryrinth sealing [2]; the determining dimensions (width of the projections, depressions, etc.) are assumed to be the same as are obtained along the trajectory of moving flow, taking into account twisting of the gas jet.

Of significant interest for quantitative and qualitative evaluation of heat and mass transfer in an air gap are the investigations of the temperature field of the air gap carried out on a special model.

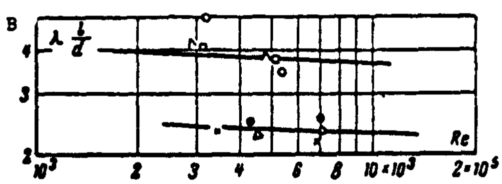

Fig. 7. Coefficients of resistance of an air gap with annular roughness of the rotor.
————) by calculation; O), ×), Δ) experimental points for $D_2 = 310$; $D_1 = 300$; $\varepsilon = 0.6$ and $n = 1000$, 2000, and 3000 rpm, respectively; O),Δ), Λ), experimental points for $D_2 = 305$; $D_1 = 300$; $\varepsilon = 0.4$ and $n = 1000$, 2000 and 3000 rpm respectively.

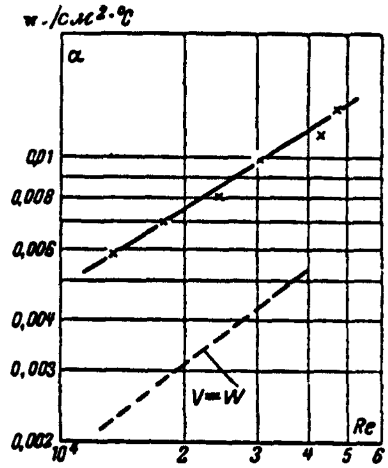

Fig. 8. Dependence of the the coefficient of heat transfer from the surface of a rotor on the rotational speed with turbulent flow. ———) experiment (when n = var and $Q = 0$); ---) calculation for equivalent velocities in a straight channel ($Nu = 0.018\ Re^{0.8}$).

It has been established that the temperature field of the air gap along the length of the machine depends on the distribution of the flow of cooling gas over the gap and the quantity of heat carried off by the gas flow. The point sources of heat or heated gases exert no noticeable local influence on the temperature field of the air gap, but they influence it only as components in the general heat balance.

Between the sections of the air gap (compartments) in which are located the sources of heated and cold gas there exists active heat and mass transfer which increases with an increase in the rotational speed and decreases with an increase of gas flow along the main channel.

The results of the investigations of heat transfer from the surface of a rotating rotor are shown in Figs. 8, 9, and 10. Figure 8 shows the experimental values of α for the surface of the rotating rotor and for a straight channel whose walls are washed at the same relative velocity as the surface of the rotor. As is evident from Fig. 8 the values of α for the surface of the rotor are approximately two times greater than in a fixed channel.

In order to explain this phenomenon let us return to an examination of the field of rotational speeds of air in the air gap of a machine [4]. The particles located closer to the rotor rotate at a higher peripheral velocity; hence centrifugal forces which are larger in magnitude than those acting on particles in the middle of the flow, forcing them back from the rotor.

Under the influence of centrifugal forces the mass transfer is intensified in the boundary layer and between the middle of the flow and the boundary layer. This same rule is confirmed by the following phenomenon which was discovered during the experiments.

Figure 9 shows the results of the experiments to determine the dependence $\alpha = f(\Delta t)$ for various rotor rotational speeds, air flow through the gaps, and temperatures of the surface of heat removed. When $\Delta t > 50^{\circ}C$ the heat-transfer coefficient α is practically independent of Δt. With small temperature drops between the air in the gap and surface of the rotating rotor the heat-transfer coefficient sharply increases when Δt decreases, i.e., with heat transfer from the rotor

the air the centrifugal forces create the reverse effect.

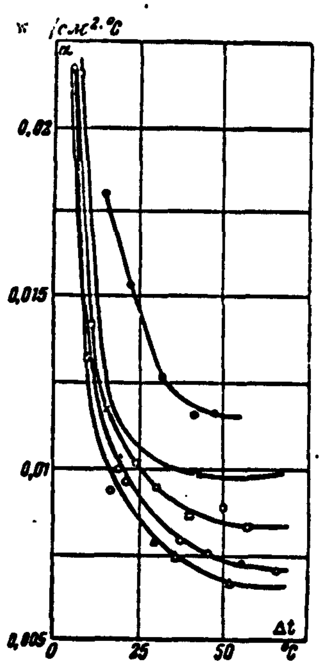

Fig. 9. Dependence of the coefficient of heat transfer from the surface of a rotating rotor on the temperature drop Q = 0.0908 m^3/sec. ●) n = 1500 rpm; ×) n = 1250 rpm; □) n = = 1000 rpm; ○) n = 750 rpm; Δ) n = 500 rpm.

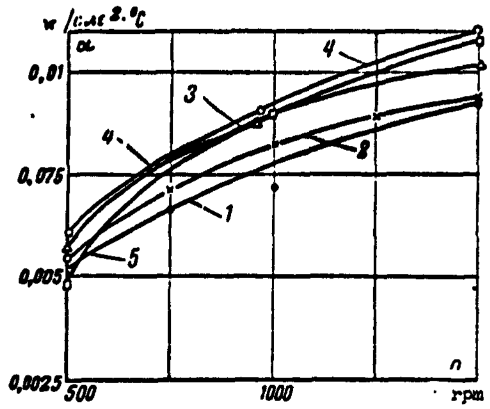

Fig. 10. Dependence of the coefficient of heat transfer from the surface of a rotating rotor α on the rotational speed and air flow through the gap. 1) when Q = 0.0895 m^3/sec; 2) Q = 0.136 m^3/sec; 3) Q = 0.182 m^3/sec; 4) Q = 0.272 m^3/sec; 5) Q = 0.

The gas particles in the boundary layer on the surface of the rotor are heated more than those in the center of the flow, and hence are lighter. The rotating rotor centrifuges the cold particles, hampering their access to the surface of the rotor and impairing heat transfer.

The phenomenon of intensifying the heat and mass transfer between the boundary layer and middle of the flow, caused by the velocity gradient in the boundary layer of the rotor, is partially neutralized.

If the axial air flow Q_a is gradually increased in the air gap with a rotating rotor, then at first with low flow rates the coefficient of heat transfer for the rotor is decreased as compared to the

regime when $Q_a = 0$; it then increases as Q_a increases. For a high flow-rate increase, α becomes larger than when $Q_a = 0$ (Fig. 10).

This phenomenon is observed both with laminar and turbulent flow. It is associated with a change in the curvature of the flow trajectory. When $Q_a = 0$ the air particles in the gap travel in a circle, when $Q_a \neq 0$ they travel in a spiral. With an increase of $Q_a \equiv v_a$ the step of the rotation trajectory becomes larger and its curvature decreases, in connection with which the angular rotational speed of the air decreases and hence the magnitude of the centrifugal forces which intensify heat exchange decreases compare curves 5 and 1); therefore at small v_a, when the velocity of air relative to the rotor $c_w = \sqrt{w^2 + v_a^2}$ practically independent of the value of v_a a reduction of α is observed with an increase of v_a.

With a further increase of v_a, there will be an increase in c_w and, naturally, the heat transfer coefficient.

When $v_a \gg w$ the effect of rotation on the value of α is imperceptible.

The results of the physical investigations on models were compared with the results of investigations on full-scale installations.

The VNIIEM [All-Union Scientific-Research Institute of the Electrical Industry] together with plants tested turbogenerators TVF-100, TVV-165, TVV-200, TVV-300, TVO-30, and TGV-200 and then made thermal and ventilating calculations.

In all cases the results of the measurements on the full-scale installations were in good agreement with the calculations for which the described results of the investigations were used as a basis.

In asynchronous machines, specialized machines, and direct-current machines good agreement between experiment and calculation is observed.

REFERENCES

1. M. A. Mikheyev. Osnovy teploperedachi, Gosenergoizdat, 1956.

2. N. Ye. Idel'chik. Spravochnik po gidravlicheskim soprotivleniyam, Gosenergoizdat, 1960.

3. V. V. Mal'tsev. Issledovaniye vnutrennego radial'nogo okhlazhdeniya rotora turbogeneratora, "Vestnik elektropromyshlenosti", 1960, No. 8.

4. V. V. Mal'tsev. Issledovaniye dvizheniya okhlazhdayushchego gaza v vozdushnom zazore elektricheskoy mashiny, "Vestnik elektropromyshlennosti", No. 6, 1959.

5. G. Taylor, Fluid Friction Between Rotating Cylinder, Proc. Roy. Soc. (A), Vol. 157, 1936.

6. S. Coldstein, .The Stability of Viscous Fluid Flow Between Rotating Cylinders. Proc. Cambr. Philas. Soc. Vol. 33, 1937.

7. A. Fage. The Influence of Wall Oscillations, Wall Rotation and Entry Eddies of the Breakdown of Laminar Flow in a Pipe, Proc. Roy. Soc. (A), Vol. 165, 1933.

8. S. Goldstein, Proc. Cambrige Phil. Soc., 33, Vol 41, 1937.

9. R. Cornish, Flow of Water Through Five Clearances With Relative Motion of the Boundaries, Proc. Roy. Soc. (A), Vol 140, 1933.

DISTRIBUTION LIST

DEPARTMENT OF DEFENSE	Nr. Copies	MAJOR AIR COMMANDS	Nr. Copies
		AFSC	
		SCFDD	1
		DDC	25
		TDBTL	5
HEADQUARTERS USAF		TDBDP	2
		AEDC (AEY)	1
AFCIN-3D2	1	AFFTC (FTY)	1
ARL (ARB)	1	ASD (ASYIM)	2
		BSD (BSF)	1
		TDES (Smith)	1
OTHER AGENCIES			
CIA	1		
NSA	6		
DIA	9		
AID	2		
OTS	2		
AEC	2		
PWS	1		
NASA	1		
ARMY (FSTC)	3		
NAVY	3		
NAFEC	1		
RAND	1		
PGE	12		